Numerical Investigation on the Performance of Inertance Tube Pulse Tube Refrigerator by Varying Compressor Amplitude

In partial fulfillment of the requirement for the degree of

Master of Technology
in

“Cryogenic & Vacuum Technology” Specialization

in

Department of Mechanical Engineering

by

Chanchal Kumar Gautam

Under the guidance of

Dr Suman Ghosh

DEPARTMENT OF MECHANICAL ENGINEERING

NATIONAL ISTITUTE OF TECHNOLOGY ROURKELA

ROURKELA-769008, ODISHA, INDIA.

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CERTIFICATE

This is to certify that the research work that has been presented in this thesis entitled "**Numerical Investigation on the Performance of Inertance Tube Pulse Tube Refrigerator by varying compressor Amplitude**" by **Mr. Chanchal Kumar Gautam**, has been carried out under my supervision in partial fulfilment of the requirements for the degree of Master of Technology in Mechanical Engineering during session 2013-2014 in the Department of Mechanical Engineering, National Institute of Technology, Rourkela.

To the best of my knowledge, this dissertation work has not been submitted in any other college or university at any time prior to this, for the award of any degree or diploma.

Date:

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DECLARATION

I certify that

a. The work contained in the thesis is original and has been done by myself under the general supervision of my supervisor(s).
b. The work has not been submitted to any other Institute for any degree or diploma.
c. I have followed the guidelines provided by the Institute in writing the thesis.
d. I have conformed to the norms and guidelines given in the Ethical Code of Conduct of the Institute.
e. Whenever I have used materials (data, theoretical analysis, and text) from other sources, I have given due credit to them by citing them in the text of the thesis and giving their details in the references.
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ACKNOWLEDGEMENT

I am extremely fortunate to be involved in an exciting and challenging research project like **“Numerical Investigation on the Performance of Inertance Tube Pulse Tube Refrigerator by varying compressor Amplitude”**. It has enriched my life, given me an opportunity to work in a new environment of multiphase flow. This project enhanced my thinking and understanding capability and after the finishing of this project, I experienced the feeling of success and satisfaction.

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Date: Signature

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NOMENCLATURE

Symbols	Descriptions
x_0	Piston displacement amplitude (m)
C_p	Specific gas constant, (J/kg-K)
C	Inertial resistance (m^{-1})
E	Total energy (JKg^{-1})
h	Enthalpy (J/kg)
k	Thermal conductivity (W/m-K)
p	Pressure (N/m^2)
T	Temperature (K)
t	Time (s)
V	Volume (m^3)
S_x, S_y	Momentum source terms (Nm^{-3})

Greek Symbols	Description
ω	Angular frequency (rad/s)
ψ	Permeability tensors (m^2)
μ	Dynamic viscosity(kg/m s)
τ	Stress tensors (N/m^2)
ξ	Porosity
ρ	Density (kg/m^3)
ν	Kinematic viscosity
τ	Stress tensors (N/m^2)

Subscripts	Descriptions
z	Frequency
x	Axial coordinate
r	Radial coordinate

ABBREVIATION

PTR	Pulse tube refrigerator
BPTR	Basic pulse tube refrigerator
OPTR	Orifice type pulse tube refrigerator
DIPTR	Double inlet pulse tube refrigerator
ITPTR	Inertance tube pulse tube refrigerator
TAPTR	Thermo-acoustic pulse tube refrigerator
CHX	Cold heat exchanger
HHX	Hot heat exchanger
AC	Alternative current
DC	Direct current

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ABSTRACT

A numerical study has been made for the predication of the performance of inertance tube pulse tube refrigerator (ITPTR). Here basically the effect of amplitude of the piston in compressor on the performance of ITPTR has been investigated. At first, a specific geometry of ITPTR with a particular compressor-amplitude (from Cha et al., 2006) has been numerically modelled and studied and the results has been compared (with Cha et al., 2006) for the validation purpose. Then another one geometry of ITPTR has been considered and the amplitudes of this ITPTR compressor piston has been varied in two different ways under the same boundary conditions. In the first case of second geometry, the amplitude doubled but the swept volume of compressor remain same as in the reference compressor (taken from Cha et al., 2006) and in the second case of second geometry, both the amplitude and volume of the compressor are doubled from the same reference compressor. The Finite Volume Method has been adopted here for all the present numerical simulations. The result shows the effect of variation of amplitude on the performance of ITPTR.

Keywords: Compressor; Amplitude, After cooler, Cold Heat Exchanger, Regenerator, Hot Heat Exchanger, Pulse tube, Inertance tube, Reservoir, (ITPTR), Finite Volume Method.

CHAPTER 1

INTRODUCTION
AND
LITERATURE SURVEY

1.1 INTRODUCTION

1.1.1 General

The cryogenics is known as “the production of freezing cold”. It is a Greek word. Cryogenics is used today as a synonym for the low-temperature condition. It is the study of the behavior and production of material at very low temperature, which is below than. 123 K. The pulse tube refrigerator is a very small and attractive device, which can reach the temperature ranges between absolute zero to 123K for cryogenic cooling. The advantage of pulse tube refrigerator is performance efficiency, low vibration, reliability, small size and weight, no moving part in cold end, long life time over the Stirling and Gifford-McMahon refrigerators. After the time of invention of pulse tube refrigeration many improvement has been done to obtain the lower cryogenic temperature and efficiency. There are lot of variation in basic pulse tube refrigerator like orifice pulse tube (OPTR), multi-stage pulse tube refrigerator, double inlet pulse tube refrigerator and in recent time inertance tube pulse tube refrigerator. The main problem in all type pulse tube refrigerator the compression and expansion of working fluid not well defined and the no clear description for understand the correct adjustment between mass flow rate and pressure to achieve high efficiency for high heat transfer. Resonance of wave in BPTR occurs due to phase phenomena. To find optimal phase adjustment the orifice of (OPTR) replaced by inertance tube (very small diameter and a long thin tube) of (ITPTR). Inertance tube has similar work as inductance in electrical system by adding this it improved efficiency of power transfer at cold region of pulse tube. In present work two ITPTR specific dimension and same boundary condition but different in volume and amplitude considered. Simulating and examine the different effect on different parts of ITPTR.

1.1.2 Applications of cryocoolers:

There are many application of the cryocooler.

Medical

In MRI cooling superconducting magnets.

Oxygen Liquefaction for hospital and home purpose.

Cryosurgery and cryogenic catheters.

For heart and brain studies by SQUID magnetometers.

Environmental

For pollution monitoring by infrared sensors.

For studies of atmospheric by infrared sensors.

Military

For guidance of missile by Infrared sensors.

Using monitoring nuclear activity by Gamma-ray sensors.

For mine sweeping by Superconducting magnets.

Energy

For peak shaving by LNG.

Superconducting power applications (motor, transport).

For thermal loss measurements by Infrared sensors.

Commercial Uses

Gas liquefaction in industrial application.

Cellular-phone base stations, voltage standards by superconductors.

Semiconductor fabrication by superconductors.

1.1.3 Classification of Cryocoolers

Cryocooler are classified into two type.

A. Recuperative type

B. Regenerative type

1.1.3.1 Recuperative type

Recuperative type cryo-cooler have a heat exchanger which is recuperative type and in this system refrigerant move in steady flow. In these type cryocooler have a compressor which have a constant pressure at inlet and outlet. There must have inlet and outlet valve in reciprocating type compressor and no need of valves in scroll, screw or centrifugal compressor for steady flow. Affinity to DC type electrical network for recuperative cryocooler. Recuperative heat exchanger has two or more than two channel. The heat exchanger also affected by the properties of refrigerant. In this system compressor end has the maximum loss of energy. It can be scaled to any size, this is great advantage of it. Brayton and JT cryocooler are example of recuperative type cryocooler.

1.1.3.2 Regenerative type

Regenerative type cryocooler must have a regenerative heat exchanger and its flow is waver flow and pressure. Affinity to AC type electrical system for regenerative heat exchanger. In this system pressure affinity to voltage and volume flow is affinity to current. These type of heat exchanger have fine-mesh screen matrixes which is very useful for improving heat transfer. In the first half of cycle heat are stored in the matrix of regenerator in forward motion and second half of cycle heat released by matrix of regenerator in same channel. It must have its initial temperature after the full one cycle. Stirling refrigerator, Gfford-McMohan refrigerator and pulse tube refrigerator is example of regenerative cryocooler.

Classification of cryocooler

Regenerative cryocooler

Gifford-Mc Mohan Pulse Tube (With valve)

Stirling Type Pulse Tube (Without valve)

Recuperative cryocooler

Joule Thomson

Brayton

Claud

1.1.4 Types of pulse tube refrigerator

- In view of geometry or shape

    1. Pulse Tube Refrigerator In line type
    2. Pulse Tube Refrigerator Coaxial type
    3. Pulse Tube Refrigerator U type

- As per improvement

    1. Basic Pulse Tube Refrigerator (BPTR)
    2. Pulse Tube Refrigerator of (OPTR) Orifice type

    3 Pulse Tube Refrigerator of Double inlet type (DIPTR)

    4 Pulse Tube Refrigerator Multi inlet type

1.1.4.1 Basic Pulse Tube Refrigerator

In early sixties initial observed by Gifford and Longsworth, the one part of hollow tube has cooled by oscillating pressure in other part. It was the beginning of the more efficient cryogenic refrigerator known as “Basic Pulse Tube Refrigerator” (BPTR). As shown in figure it has no moving part at cold temperature part, simplicity and enhanced reliability because of these properties it is most important topic in cryogenic refrigerator.

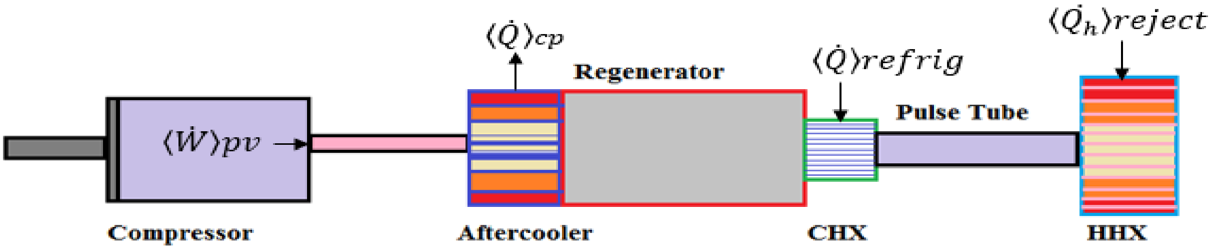


Figure 1.1: Block diagram of basic pulse tube refrigerator

1.1.4.1.1 Main component of pulse tube refrigerator

1. **Compressor: -** The core objective of compressor in closed ch amber is to generate the pressure for refrigerant and producing a harmonic oscillation for the gas inside the system.

2. **After cooler: -** It extract the heat from compressor and dispose to the environment.

3. **Regenerator: -** It has the porous media matrix which is working like capacitor. It takes the heat in first cycle and release the heat in second cycle. In regenerator should be no pressure drop and 100% effectiveness to attain the maximum flow enthalpy in the system of pulse tube.

4. **Cold Heat Exchanger: -** It work is same as vapour compression cycle of evaporator It absorb the heat load which chamber want to get cryogenic temperature.

5. **Pulse Tube: -**This is very important part of the pulse tube refrigerator. By correct phase

shifting the heat flow of the cold hollow part to warm hollow part by enthalpy flow. The very important role for heat supply from CHH to HHX is correct phase shifting between mass flow and pressure.

6. **Hot Heat Exchanger: -** It dispose heat of compressor to environment in every periodic cycle. It also extract the heat in form of enthalpy by pulse tube reject throw environment.

7. **Orifice Valve, Inertance tube: -** The main objective of the orifice valve or inertance tube is to correct adjustment between mass flow and pressure.

8. **Buffer: -** It is a reservoir which has more volume in compare with the rest of system.

1.1.4.1.2 Working principal of pulse tube refrigerator

There are six different part of basic pulse tube refrigerator compressor, after cooler, CHX, pulse tube and HHX. First refrigerant fluid oscillate by compressor in the hollow chamber after this the heat extract form after cooler heat exchanger which are produced by the oscillating compressor. The refrigerant enter in regenerator heat are extract in first cycle and heat release in second cycle to achieve mode of steady periodic mode. The CHX is the storage which want to cool at cryogenic temperature and this heat absorb from the chamber and release to environment. After it by enthalpy pulse tube pass along the heat from cold part to the warm part. A higher temperature dispose which comes receive enthalpy by HHX. The measure of high temperature they can uproot is restricted by own size and force used to drive them.

1.1.4.2 Orifice type pulse tube refrigerator (OPTR)

Pulse Tube Refrigerator of basic type are uses surface heat pumping technique. Which is not enough to reach very cold temperature. The design improved by putting the orifice after heat exchanger and adding orifice before reservoir in pulse tube. As shows in figure the Orifice type pulse tube refrigerator is advantage for improving phase correlation between mass rate flow and pressure within the pulse tube to improve heat transport phenomena.

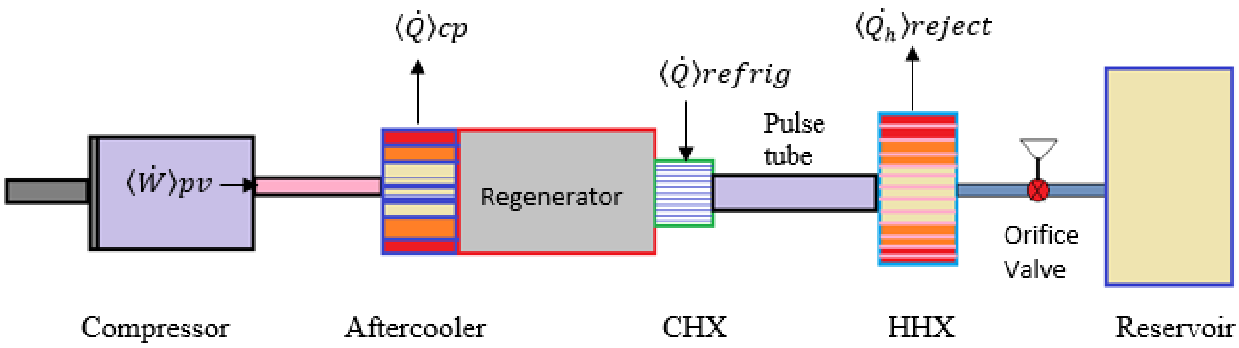


Figure 1.2: Block diagram of orifice pulse tube refrigerator.

1.1.4.3 Double inlet pulse tube refrigerator (DIPTR)

Pulse tube refrigerator efficiency can be improved by increasing by refrigeration power per unit mass rate. In this type a direct secondary orifice are connected between the warm region of refrigerator and hot region of the pulse tube. Increasing abasement of regenerator performance by mass flow throw regenerator in OPTR are improved by double inlet. As shown in figure, It has bypass for regenerator and thus reduce the power of cooling because of this the overall performance are improved.

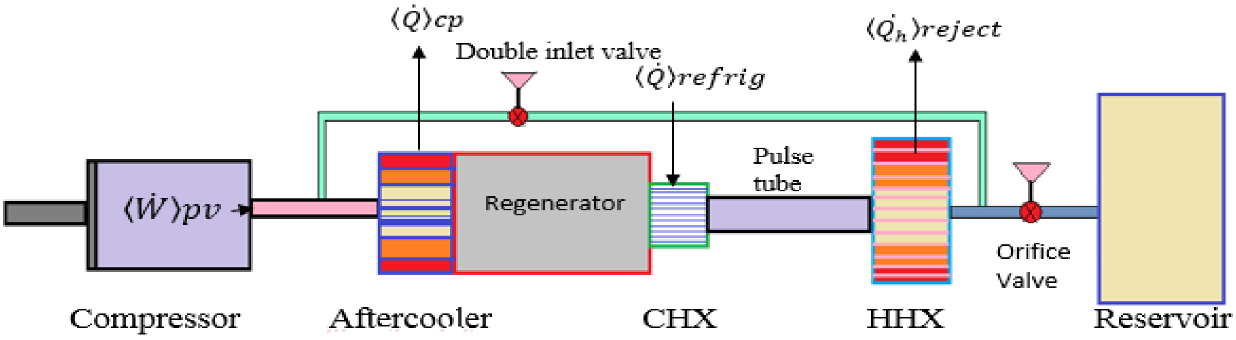


Figure 1.3: Block diagram of orifice pulse tube refrigerator

1.1.4.4 Inertance type pulse tube refrigerator (ITPTR)

The inertance tube pulse tube refrigerator is newly invention in study of PTR as shown in figure. These type of pulse tube orifice are replaced by very small diameter and long inertance tube and also added reactive impedance to the system. The correct phase shift in pulse tube and increasing flow of enthalpy by implementation of this inductance generates. These type of pulse tube refrigerator favorable for greater frequencies for higher scale pulse tube refrigerator.

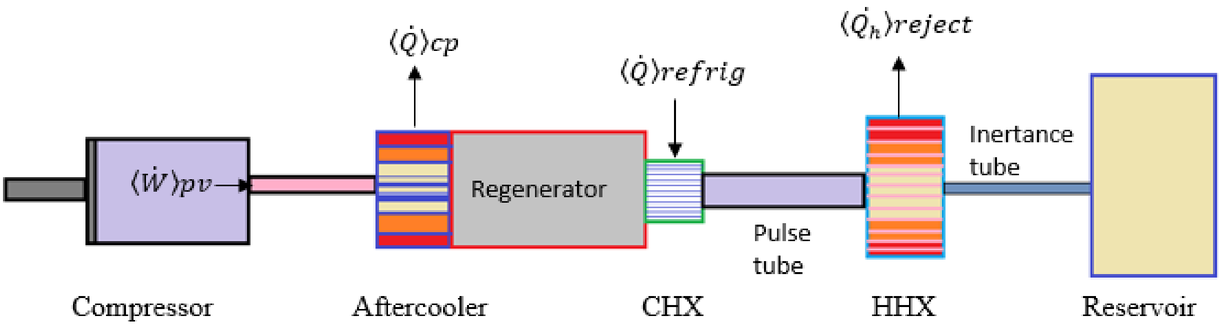


Figure 1.4: Block diagram of inertance pulse tube refrigerator

1.1.4.5 Multistage pulse tube refrigerator

It is exceptionally troublesome to accomplish cryogenic temperature by a single stage of PTR because of this using pre cooling a PTR by the other PTR. It is advantage to split into two part that is called two stag, for temperatures below 30K as shown in figure 5. It has two or more than two regenerator. After the first stage regenerator linked to the second stage regenerator. As shown in figure the heat already carry by the first stage pulse tube its remaining heat carried by the second stage pulse tube by this arrangement cryogenic temperature can achieved. As the double stage we can also arrange the more than two stage like three stage for 1.3K.

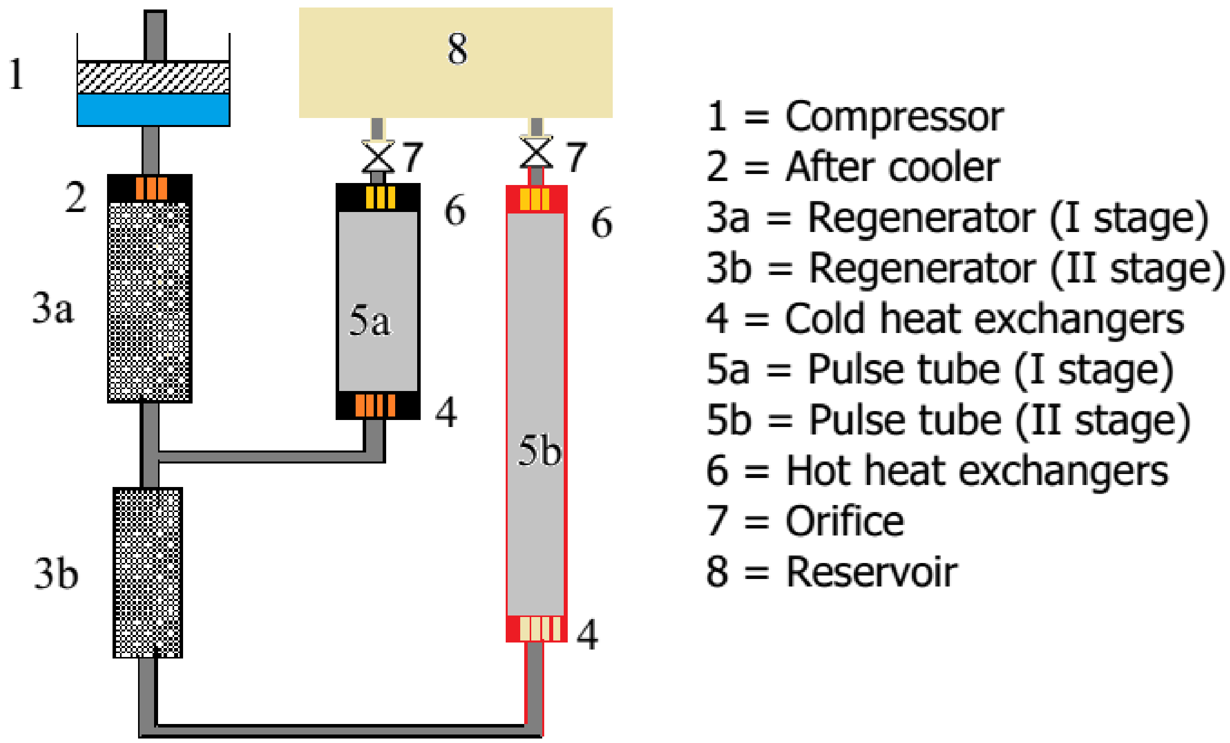


Figure 1.5: Block diagram for second stage Stirling type pulse tube refrigerator

1.2 Literature Survey

Antao and Farouk [1] investigated the hydrodynamics and heat transfer methods of a pulse tube refrigerator with orifice. An OPTR consist of the buffer, orifice valve, hot heat exchanger cold, heat exchanger, pulse tube the compressor, transfer line, after cooler, regenerator. The orifice type pulse refrigerator having helium as a working fluid, is fixed through a moving piston and is driven cyclically. The heat exchanger and regenerator are designed like a porous medium and a temperature gradient is applied into it. Different operating frequencies are set for the driver piston in order to study the function of OPTR. Due to the fluctuation of the piston and the existence of inertance tube steady- flow pattern is developed in the pulse tube, and the OPTR is affected by the expected secondary- flow recirculation pattern in the pulse tube. But when secondary flow pattern is generated in a good condition in the pulse jet, they improve the performance of OPTR and also create at the middle a zone of 'thermal buffer".

Ashwin et al., [2] studied the modeling and performance of the different types of Orifice Pulse Tube Refrigerator (OPTR) and the Inertance tube Pulse Tube Refrigerator (IPTR) with co-axial and inline configuration and compared the result. An analysis has been done on the component by the flow and heat transfer using ANSYS, and the parametric studies were performed with varying length-to-diameter ratios. The heat exchanger and regenerator are designed like a porous medium and a temperature gradient is applied into. It is found that the no-load temperature is reached in the CHX. The result also shows that non-thermal equilibrium analysis yields a lower cold heat exchanger temperature.

Ling et al., [3] studied the thermal cycle present in an inertance tube pulse tube refrigerator using CFD method. The outcome of the simulation shows as the gas pass from the ITPTR components, different thermodynamics processes takes place, and the gas parcels working on the same component under different frequencies of piston moving shows almost same thermodynamics result. Result of simulation are hence, compared with the analyzed thermodynamic result. The comparison shows that the same type types of thermodynamics cycle is generated at the same location having different frequencies and but same gas parcels.

Boer et al., [4] have done an experiment with inertance pulse tube for performance analysis. The rate of refrigeration in the IPTR are depend on the conductance of the regenerator, the driving pressure, the capacity of pulse tube, and the frequency. In the inertance tube, effective conductance is determined using a simple turbulent flow model. The result shows that inertance pulse tube (IPTR) is better than that of orifice pulse tube refrigerator in a certain range of frequencies. The result thus obtained helps in the improvement of the regenerator and pulse tube, pressure amplitude, phase angle between these parameter. The result also showed how this work is useful in experimental works.

Farouk and Antao, [5] did Numerical analysis on OPTR. These investigations consist of the hydrodynamics and thermal analysis in pulse tube refrigerator of single stage orifice type, and is moves by piston and helium as refrigerant. The regenerator and the heat exchanger are porous and temperature gradient is applied on it. Due to the inertance tube and the fluctuation in the piston develops a steady-flow pattern. The secondary flow parameter help to isolate the hot and the cold ends to isolate.

Pierre and Babo, [6] analyzed the performance of Orifice pulse tube through an perfect model, a time dependent model taking consideration of heat transfer resistance and mass transfer resistance, The effect shows that the refrigeration load is due to the entropy flow in an Ideal model, which clarify minimum entropy generation in the Orifice tube and a maximum Co-efficient of Performance, Further the dynamic model obtained from the simulation is compared from experimental data which is obtained from Helium and Argon., in which the indicated diagram shows that how the energy exchange takes place and also help s in efficient design for Pulse tube refrigerator.

Yan et al., [7] did an experiment on 15 K two-stage Stirling-type pulse-tube cryocooler, which is run by a linear compressor. The compressor consist of dual-opposed-piston, a moving-magnet having a plate spring. The two-stage cold head of the pulse-type cryocooler is separated by a gas. Cryocooler having the phase shifter can be a double-inlet or an inertance-tube type. The result obtained from the experiments shows that a pressure ratio of 1.3–1.5 is obtained due to linear compressor. The result also shows that when a charging pressure of 1.2 MPa and a frequency of 32 Hz is applied, the cooling temperature of the 1^{st} and 2^{nd} stage cryocooler are 93.3 K and 14.2 K respectively.

Chen et al ., [8] have perform an experimental investigation on a two-stage pulse tube refrigerator of double-orifice which consists of a regenerators, helium compressor, double-orifice , pulse tubes reservoirs and a heat exchangers. The new phase shifter i.e. double-orifice is located at room temperature. A double-orifice is introduced to function the pulse tube from room temperature to temperature of liquid helium. The oscillating pressure in a pulse tube refrigerator is generated by a helium compressor. The input power of helium compressor 3.7 kW. Timing is controlled by a solenoids and computer. The experimental result shows that, the smaller orifice has much effect on the performance of cooling. The result also shows that with the introduction of double-orifice, a lowest of 3.1 k temperature is obtained.

Farouk and Antao, [9] Investigated the characteristics of transport phenomenon and estimate the performance of an orifice pulse tube refrigerator (OPTR) by numerical model. There describe two type of OPTR. For case-1 only change the pulse tube taper angle and for case-2 change the pulse tube taper angle as well as change hot heat exchanger diameter. The satisfactory result show the effect of the taper angle of pulse tube on secondary flow form in the pulse tube. Only case-2 shows reduction of discharge of velocity flow at hot end by tapering the pulse tube by this developed the performance of OPTR.

Cha et al.,[10] Investigate the two mode pulse tube refrigerator of inertance tube type with compelled by different dimensions of a compressor, an after-cooler, a regenerator, CHX, pulse tube, HHX containing different boundary condition modeled and simulated. The result shows the complex model also simulate easily in CFD model. There must be greater length-to-diameter ratio for one dimension flow model. The multi-dimensional flow are appear at the intersection of two components and generation of secondary recirculation flow occur where ratio of length-to-diameter ratio are relatively very small.

Sachindra et al., [11] there are two methodology Non-Sorted Genetic Algorithm II (NSGA II) and Response Surface Methodology (RSM) are used to dimension optimization of regenerator and pulse tube. In designing Box-Behnken used surface method with help of two level and four factor. In this designing only length and diameter of regenerator and pulse tube changed and other part is same. Here in this method input is compressor power and output the temperature (T_{cold}) of CHX. ITPTR modeled and sold with help of CFD. The result shows the investigation of independent variable effect on responses with RSM. The (ANOVA) method are used to investigate the model. Non-sorted genetic algorithm II (NSGA-II) also used in study and optimization of RSM model.

Zhang et al., [12] in this paper have to take a simple orifice pulse tube refrigerator for modeling and simulation by CFD. The result shows the phase-leg between mass flow rate and pressure in CHX and also shows the oscillation of temperature with respect to different load. In this investigate effect of oscillation pressure in pulse tube. The all result are validate by the modeling. The poor overall efficiency due to the swirl in mixing as unsteady in CHX.

Tao et al., [13] here in this paper introduce both 1-D and 2-D model together for better understand nature of heat transfer and fluid flow for basic type, double-inlet type and orifice type pulse tube refrigerators. The computational time greatly reduced to analyze the fluid flow hand heat transfer and also the numerical approach can be done for all three pulse tube refrigerator. The result shows the temperature annular effect, velocity and DC flow reason for complicated heat transfer and fluid flow. The result also give the better approach to understand the process of thermodynamic for PTRs.

Srinivas et al., [14] at no load condition CHX temperature cool down 49.9 k with the 3.5 MPa average pressure and 120 Hz and in 5.5 min CHX achieve the 80 K. the result shows the net refrigerant power 3.35 W at 80 K and efficiency of Carnot engine is 19.7%. This is first time to achieve such a lower temperature at the CHX at above 100Hz. Thus for given power reduced the volume of cryocooler operating at high frequency. It used in development of microelectrode mechanical system by help of micro cryocoolers.

Ghahremani et al., [15] investigated the effect of the high capacity pulse tube refrigerator which provide cryogenic temperature more at than 250 W. In present case study the effect of the geometry design and operating parameter for pulse tube refrigerator of double inlet (DIPTR) then compare it type pulse tube refrigerator. The present model compare with the experimental data for validation. The result also shows that the optimum HCPTR. It proposed that at 50 Hz and load 335 W obtained the 80 K CHX temperature and COP of 0.05.

1.2.1 Gaps in the Literature

The above discussion gives a broad overview of the multiple research activities carried out so for to analysis of the pulse tube refrigerator. But yet many scopes of research on the pulse tube refrigerator By the Literature review we can suppose that there are many things to do the rest. There are some gaps in the literature survey.

1. It is very difficult to evaluate the optimum parameters of PTCs by experimental method. It is also very difficult developing code for CFD analysis.
2. To overcome above problem can use CFD for simulating model and testing analysis which carry out less time and capital

1.3 Aims and Objectives

The basic goal of the present investigation to study the effect of the different dimension of compressor on the nature of temperature profile of different parts of the pulse tube mainly on cold heat exchanger and also study what is effect of pressure on the different part of inertance pulse tube pump. Here picked Helium as working fluid on the grounds that it has most reduced critical temperature contrast with different gasses and additionally high thermal conductivity.

1.4 Organization of the Thesis

The dissertation is split in five sections. In first chapter consist introduction part, which has been explain about topic and literature survey. After that gap in the literature has been defined. Then direct and aim has been excused. Chapter 2. Defined the problem statement, it discussed about the problem description in schematic form. Chapter 3. Developed the methodologies to solve the problem. Chapter 5. Clearly mention about the significant conclusion and future orbit of the project.

CHAPTER 2

MODELING

2. MODELING

In this chapter, detailed description of the problem in the contour of the schematic diagram. The different cases have been detail discussed related to the problem.

2.1 Modeling of inertance type pulse tube refrigerator

Now a days the inertance pulse tube refrigerator are instead of the orifice valve. The advantage of this correct phase shift in pulse tube and increasing flow of enthalpy by implementation of this inductance generates. Already discussed about it above.

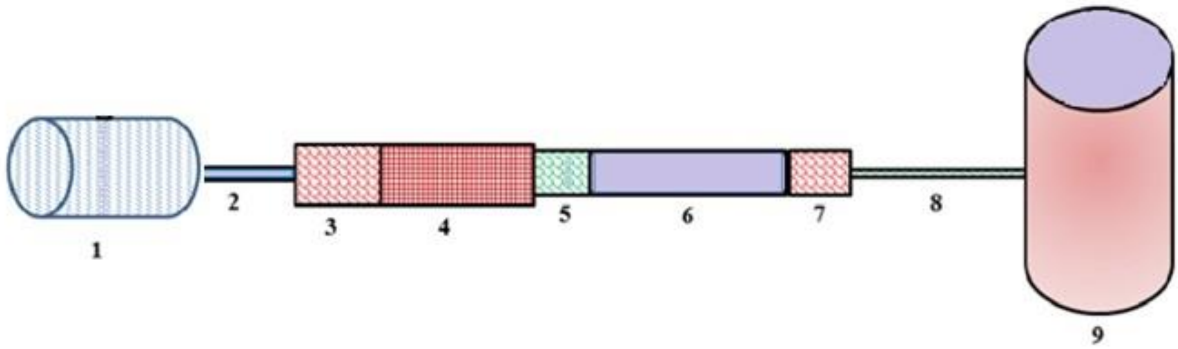


Figure 2.2: Block Diagram of ITPTR: 1-Compressor, 2-Transfer line, 3-After Cooler, 4-Regenerator, 5-CHX, 6-Pulse tube, 7-HHX, Inertance Tube, 9- Reservoir.

2.2 Problem Description

Computational fluid dynamics, typically truncated as CFD, is an extension of fluid mechanics that uses numerical strategies and calculations to tackle and break down problem that include fluid streams. Computer are utilized to perform the counts needed to reproduce the communication of fluids and gasses with surfaces characterized by boundary conditions. With fast supercomputers, better results might be attained. Continuous examination yields programming that enhances the correctness and velocity of complex reproduction situations, for example, transonic or turbulent streams. ANSYS, ZEUS-MP, GADGET, FLASH, Open FOAM, STAR-CD and FLOW3D tool are the best example of CFD tool. The ANSYS is most useful for CFD codes. ANSYS is very useful to solve the problem related to fluid flow and flow of heat process in very complicated engineering problem. With the help of ANSYS desiccate code and using the (UDF) for dynamic boundary condition. The model in which contain volume compression and expansion can create deforming mesh volume by the function of dynamic meshing function. So the ANSYS has the credential to solve the dynamic type volume problem, creating UDF boundary condition and modeling credential for porous media. That's why it chooses to solve the ITPTR, Stirling type ITPTR.

The pressure wave is directly generated by a valve less compressor in a pulse tube refrigerator of Stirling-type. A Stirling type-pulse tube refrigerator used for higher frequencies like 34 HZ. The all dimension of present model re-taken from the literature for (ITPTR). Fig.7 explain the axis-symmetric 2-dimensional co-ordinate system.

The geometric modeling and meshing of various parts of inertance pulse tube refrigerator has been done using Fluid Flow (ANSYS). The model of ITPTR was developed 2D axisymmetric because symmetry of model. For meshing the model quadrilateral and triangular meshes are used. The important parts of model are Compressor, Transfer line, After-cooler, Regenerator, CHX, Pulse tube, HHX, Inertance tube, Reservoir. There are only one part is movable and other parts are in rest. The all dimension are given below in Table.1 and Table.2.

Case 1. The amplitude doubled but the swept volume of compressor remain same as a reference compressor i.e. taken from Cha et al. (2006).

Table 2.1: All dimension of different parts of IPTPR with dimension 1, case 1

. Components	Radius (m)	Length (m)
(1) Compressor	0.006700	0.01500
(2) Transfer-line	0.001550	0.11600
(3) After-cooler	0.004000	0.13600
(4) Regenerator	0.010000	0.16100
(5) Cold heat exchanger	0.003000	0.16670
(6) Pulse tube	0.007500	0.19670
(7) Hot heat exchanger	0.004000	0.20670
(8) Inertance tube	0.000596	1.27670
(9) Reservoir	0.013000	1.40670

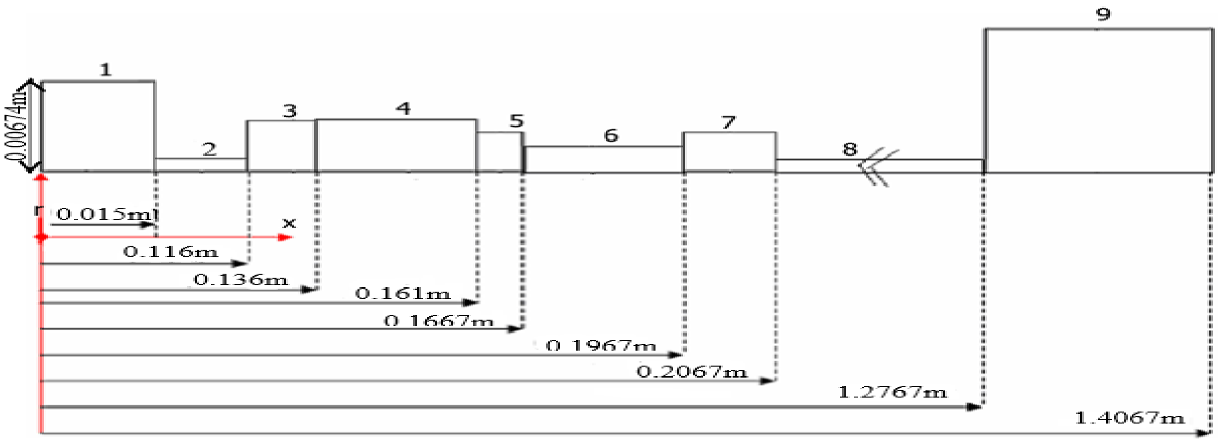


Figure 2.3: 2-D axis-symmetric geometry of ITPTR

Case 2. The both the amplitude and volume of the compressor are doubled from the same reference compressor i.e. taken from Cha et al. (2006).

Table 2.2: All dimension of different parts of IPTPR with dimension 1, case 2.

Components	Radius(m)	Length(m)
(1) Compressor	0.009540	0.01500
(2) Transfer-line	0.001550	0.11600
(3) After-cooler	0.004000	0.13600
(4) regenerator	0.010000	0.16100
(5) Cold heat exchanger	0.003000	0.16670
(6) Pulse tube	0.007500	0.19670
(7) Hot heat exchanger	0.004000	0.20670
(8) Inertance tube	0.000596	1.27670
(9) Reservoir	0.013000	1.40670

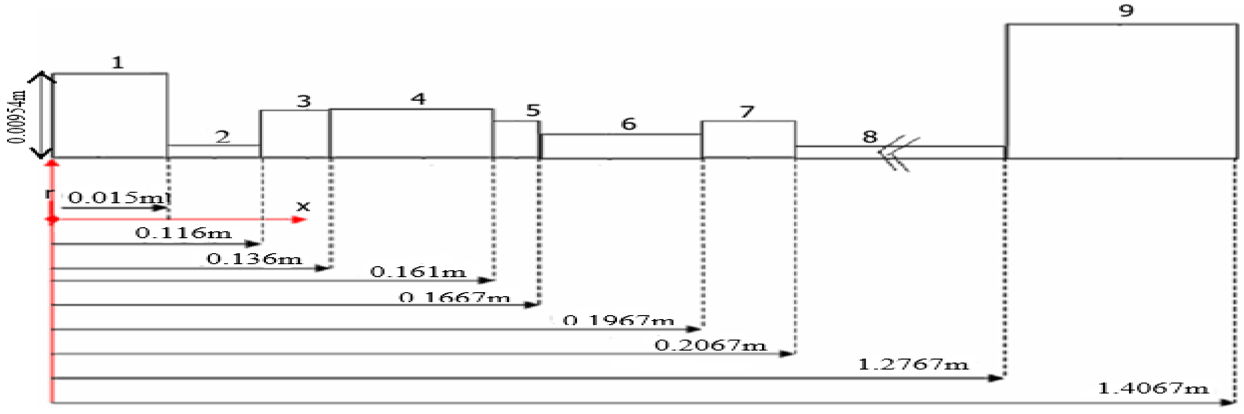
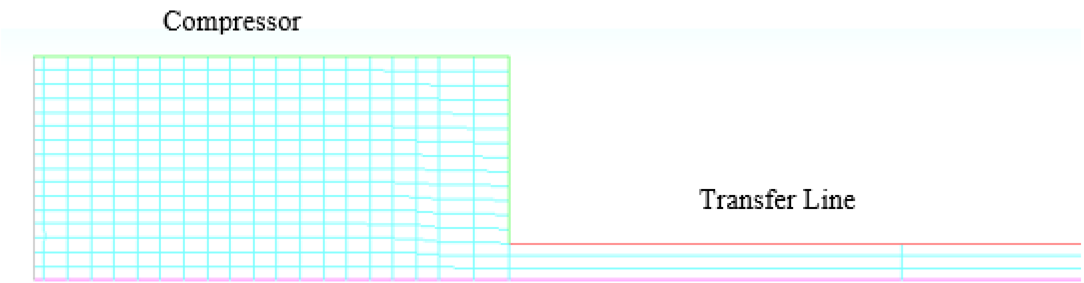
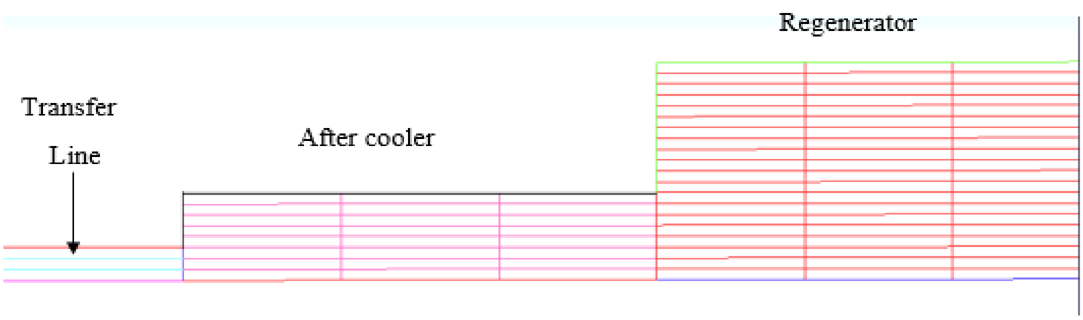


Figure 2.4: 2-D axis-symmetric geometry of ITPTR

First of all geometry is created individually after this all component geometry compelled and form a system. To enable different boundary condition for different parts of individual geometry of the system. Before initiating meshing process first faces or volume are created. A partial differential equation creates an infinite dimensional space which is not a favorable condition for solution it must be in finite dimension space because of this creating mesh.

Triangular elements, quadrihedral elements and hexahedral elements. Quad elements could be used for the meshing if there is no any complex structure for computation domain. The complete meshing of axis-symmetric geometry of different zones of ITPTR. After completed meshing it exported to ANSYS for define the boundaries.





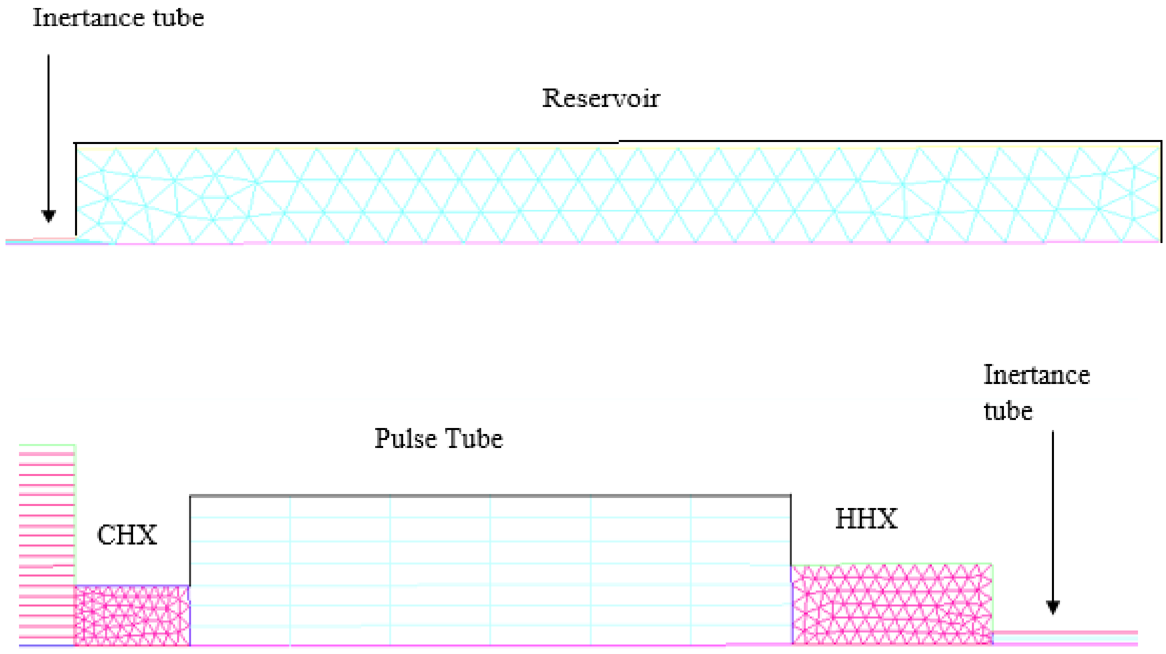


Figure 2.5: 2-D axis-symmetric meshing for case-1

CHAPTER 3

PROBLEM FORMULATION

3 PROBLEM FORMULATION

3.1 Numerical Calculation

The finite-volume method (FVM) is a technic for constitute and estimation partial differential equations in form of algebraic form of equation. It mostly used to solve governing equations for mass and heat transfer and fluid flow problem. It has advantage to create to allow for unstructured meshes. The most propelling characteristics of the FVM are that the ensuing result fulfills the conservation of quantities, for example, mass, energy, momentum and species. It contained for the whole computational domain as well as any type control volume. It shows the exact integral balance for a coarse grid. It also used for complex structure or any type grids like (Body fitted or Cartesian, structured or unstructured).

Thus to solve fluid flow and heat transfer using ANSYS, CFX Star-CD etc as commercial packages. In this method, the result space is subdivided into consistent cells or control volumes where the variable of diversions is spotted at the centroid of the control volume structuring a grid.

The following steps is to coordinate the differential type of the overseeing mathematical statements over each one control volume. Introduction profiles are then expected keeping in mind the end goal to depict the variety of the concerned variables between cell centroids. There are a few plans that might be utilized for addition, e.g. central differencing, upwind differencing, energy-law differencing and quadratic upwind differencing plans. The emanate equation is called discretization equation. Thus the discretization equation statement communicates the preservation standard for the change inside the control volume. These variables structure a set of algebraic equations which are explained at the same time utilizing extraordinary calculation.

3.2 Governing Equation

The governing equations like continuity, momentum equation on both radial and axial, energy equation for solid porous written with no swish assumption. To calculate momentum losses there are two excess source term should be known for both axial and radial direction suppose for porous zone. Solid matrix of porous zone is suppose as homogeneous and for other porous medium zone porous value are assumed to be zero.

After cooler, CHX, HHX and regenerator all four has porous media and mass, momentum, energy equation mentioned below.

$$\frac{\partial}{\partial t}\left[\varepsilon\rho_f\right]+\frac{1}{r}\frac{\partial}{\partial r}\left[\varepsilon r\rho_f\nu_r\right]+\frac{\partial}{\partial x}\left[\varepsilon\rho_f\nu_x\right]=0 \qquad (1)$$

$$\frac{\partial}{\partial t}\left[\varepsilon\rho_f\vec{v}\right]+\nabla.\left[\varepsilon\rho_f\vec{v}\vec{v}\right]=-\varepsilon\nabla p+\nabla.\,[\varepsilon\overline{\overline{\tau}}]-[\mu\overline{\overline{\beta^{-1}}}.\vec{\jmath}+\frac{1}{2}\overline{\overline{C}}\rho_f.\,|\vec{\jmath}|\vec{\jmath}] \qquad (2)$$

$$\frac{\partial}{\partial t}\left[\varepsilon\rho_f E_f+(1-\varepsilon)\rho_s E_s\right]+\nabla.\left[\vec{v}\left(\rho_f E_f+p\right)\right]=\nabla.\left[\left(\varepsilon k_f+(1-\varepsilon)k_s\right)\nabla\mathrm{T}+(\varepsilon\overline{\overline{\tau}}.\vec{v})\right] \qquad (3)$$

Assumed data based experiments

$\varepsilon = o.69, \overline{\overline{\beta}} = 1.06e \times 1o^{-10}\ m^2$, $\overline{\overline{C}} = 7.609 \times 1o^{4}\ m^{-1}$

Mass, momentum, energy equation for rest other components given below.

$$\frac{\partial\rho_f}{\partial t}+\frac{1}{r}\frac{\partial}{\partial r}\left[r\rho_f\nu_r\right]+\frac{\partial}{\partial x}\left[\rho_f\nu_x\right]=0 \qquad (4)$$

$$\frac{\partial}{\partial t}[\rho_f\,\vec{v}]+\nabla.\left[\rho_f\vec{v}\vec{v}\right]=-\nabla p+\nabla.\,[\overline{\overline{\tau}}] \qquad (5)$$

$$\frac{\partial}{\partial t}\left[\rho_f E\right]+\nabla.\left[\vec{v}\left(\rho_f E+p\right)\right]=\nabla.\left[k_f\nabla T+(\overline{\overline{\tau}}.\vec{v})\right]\frac{\partial}{\partial t}\left[\varepsilon\rho_f\right]+\frac{1}{r}\frac{\partial}{\partial r}\left[\varepsilon r\rho_f\nu_r\right]+\frac{\partial}{\partial x}\left[\varepsilon\rho_f\nu_x\right]=0$$

Where $E = h - \frac{P}{\rho} + \frac{\nu^2}{2}$

r = Radial coordinate, x = Axial coordinate, ν_r = Velocity in radial coordinate

ν_x= Velocity in axial coordinate

3.3 Steps involved in CFD problem

- According to the given flow condition the physical domain are identified.
- There are different boundary condition, properties of fluid specified and selected convenient transport equation.
- To form a grid there nodes are specified in space and each node defined by the control volume.
- To convert integral form to algebraic for a control volume integrated by differential equations. There is most discretization method is finite volume and finite difference method.
- To solve the algebraic equation by a numerical method and to execute the numerical method by a developed computer program.
- The solution is explained
- The analysis of post process is done by result displayed.

3.4 Specify the model

Before next step iterating, in the ANSYS must be define the solver, thermal and fluid properties and material to solve the fluid problem by define the boundary conditions and operation conditions for model. There are different condition define for model which is given below.

- **Solver**

  Implicit formulation, unsteady time, segregated solver, unsteady time.

- **Energy and Viscous equation**

  Selected energy equation and K-epsilon set of equation

- **Material**

  Working fluid taking as Helium (ideal gas) and frame material copper and steel.

- **Operation condition**

101325 Pa was chosen for operating pressure.

- **Boundary condition**

Hot heat exchanger and after-cooler are choses isothermal walls and others walls choses adiabatic wall with no heat flux as shown in table. The porous medium are given for regenerator.

Table 3.1: Table for initial and boundary condition for ITPTR

Case study	Boundary condition	Material
Compressor wall	Adiabatic wall	Steel
Transfer line wall	Adiabatic wall	Steel
After cooler wall	293K	Copper
Regenerator wall	Adiabatic wall	Steel
Cold end wall	Adiabatic wall	Copper
Pulse tube wall	Adiabatic wall	Steel
Hot end wall	293K	Copper
Inertance tube wall	Adiabatic wall	Steel
Surge volume wall	Adiabatic wall	Steel
Viscous resistance(m^{-2})	9.44e+9	
Inertance resistance(m^{-1})	76090	
Initial condition	300K	
Cold end load	0W	

- **Limits**

Pressure 30 to 34atm.

- **Relaxation factors**

Energy- 0.8, Momentum- 0.4, Pressure- 0.2.

- **Discretization:**

Momentum, density, energy are choses as First Order windward.

Energy chose to be 1e-06 and continuity convergence criterion, y-velocity, x-velocity, k are choses 0.001.

Defining the Material properties:

In the project cases chosen the working fluid as Helium and solid as copper and steel. Density, viscosity, diffusivity, thermal conductivity and specific heat these properties can be defined.

Defining the Operation condition:

In this pressure and gravity is operation condition. There is no important role of gravity effect because of model is horizontal axis-symmetric. Here operation pressure is set 1 atm. In term of absolute.

Defining the Porous zone:

Using porous media methods for after cooler regenerator, after cooler cold and hot heat exchanger of pulse tube refrigerator. It has special fluid zone. Porous zone option in fluid panel is permitted to indicate fluid region is a porous region. Expanding the panel shows media inputs.

There are porous media which used user inputs are:

- Specify the porous zone.
- Define the fluid flowing material through porous media.
- Define solid material for porous media.
- Set the inertial resistance coefficient, viscous resistance coefficient, and also specify the direction vector for application.

3.5 Dynamic meshing function

The dynamic meshing function are used in ANSYS. The mainly dynamic mesh model in ANSYS are used where the domain shape is not constant it vary with time, the shape of the domain always change like in reciprocating compressor there are the no fix boundaries it change with piston moving forward and backward motion as shown in fig.8. These type can be solve by using user defined meshing function. There are both type of motion can be handle which recommend or not recommend motion. Where the solution of eventually motion is insistent on the solution at current time. In the ANSYS automatic handled volume mesh based on new state of boundaries at each time step. It must be define a starting volume mesh and representation motion of moving region by using dynamic mesh model. By using user defined functions (UDFs) ANSYS can explain the dynamic motion boundary.

Because of versatility of ANSYS, it is efficient to solve the dynamic mesh model like compression and expansion of the compressor in pulse tube refrigerators. Remeshing, layering and smoothing these are different method available in ANSYS. The compressor wall modelled solid wall, along the solid wall it move for sinusoidal oscillates. Normally adiabatic boundary specified for cylinder and piston walls. The work input provided by the piston in the compressor cylinder for drive the cycle by oscillating pressure. The ANSYS dynamic function are used for modeling of the piston and cylinder. To simulate the piston cylinder effect C programming language developed by user defined function (UDF).

The mesh motion shows direction of motion of the mesh from its initial position with respect to time. Compression and expansion process of compressor easily shows.

Initial position Compression process Expansion process

Figure 3.1: Compressor and Expansion of dynamic meshing

CHAPTER 4

RESULT AND DISCUSSION

4. RESULT AND DISCUSSION

4.1 Results for Geometry-1(Validation)

To check the accuracy of the present model and the method of solution against preceding published cryogenic journal by Cha et al. [10]. It is absorb from the figure 10, that for both cases at 83 sec the present model reporting a temperature of 95 K using 6360 members of cells and in the Cha et al. (2006) case it is found 98 K using 4200 Cells

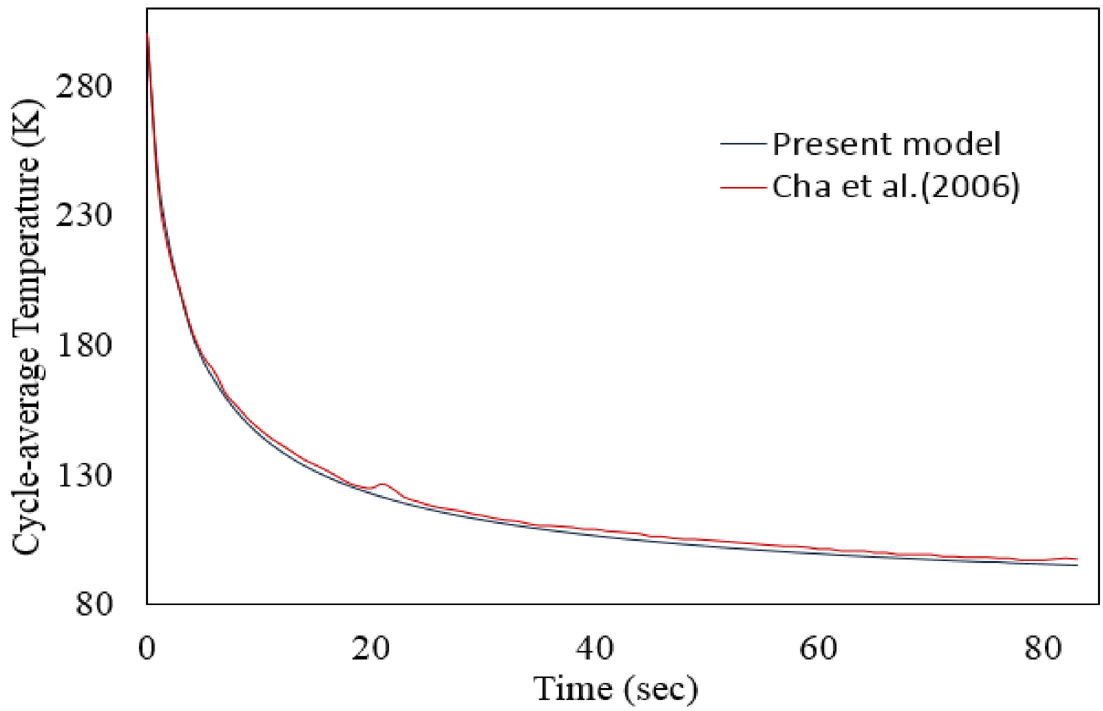


Figure 4.1: Validation plot between present model and Cha et al

Figure 4.1 shows the temperature variation with respect to the time for cha et al. (2006) model and present model. From this figure it is clear that the results for the present model perfectly machine Cha et al. (2006). That is the rate of cooling are same for two cases. The difference in temperature for these two cases after time after 83 second is 3 K only.

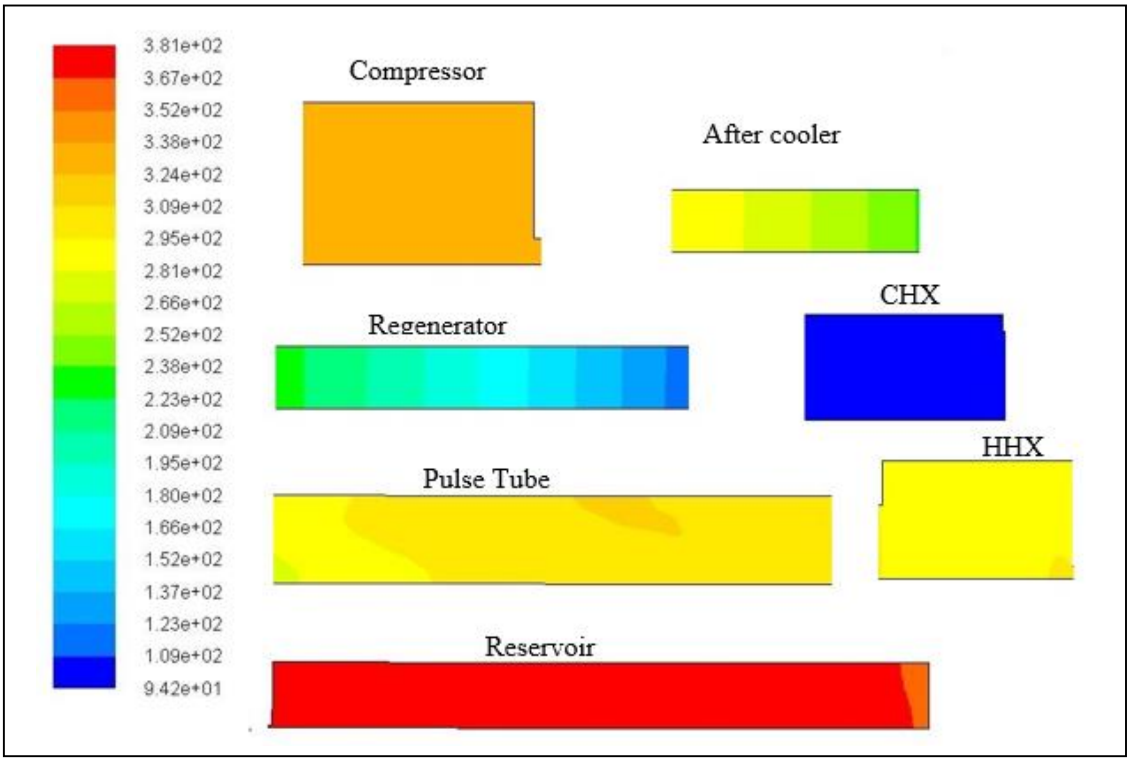


Figure 4.2: Temperature contour using present model with geometry 1at 83 second.

The Figure 4.2 shows the temperature contour for different parts of ITPTR with geometry-1 using present model. The dynamic meshing of the compressor and meshing of whole component of (ITPTR) and solution method is accurately done. Thus the present model can be selected as a reference for the solution method and modeling of other models.

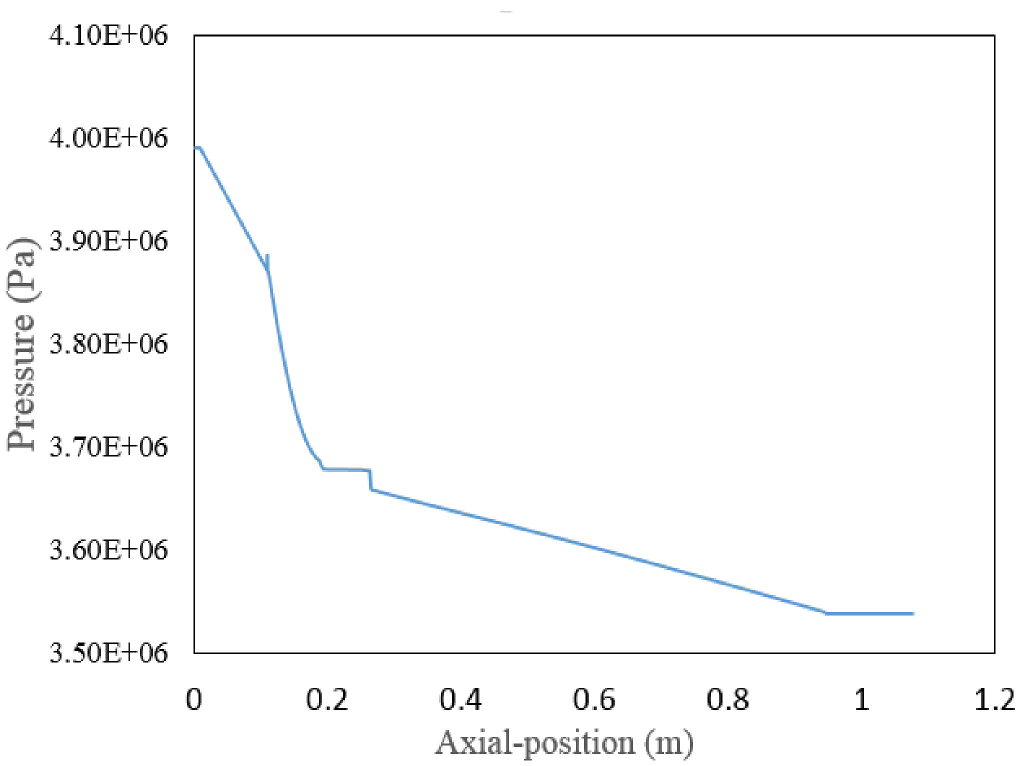


Figure 4.3: Axial Pressure variation from compressor to reservoir at the 83 second.

Above Figure 4.3 shows that axial variation of the pressure at 20 second. Here the initial pressure is 4 MPa by changing the axial position the pressure also changes the highest pressure at the compressor is 4 MPa ad it decreases rapidly with position at cold heat exchanger it shows the constant pressure and after that it decreases up to inertance tube and after that at 3.55 MPa found at the reservoir.

4.2 Results for different cases

4.2.1 Case-1: When the amplitude doubled but the swept volume of compressor remain same as a reference compressor i.e. taken from Cha et al. (2006).

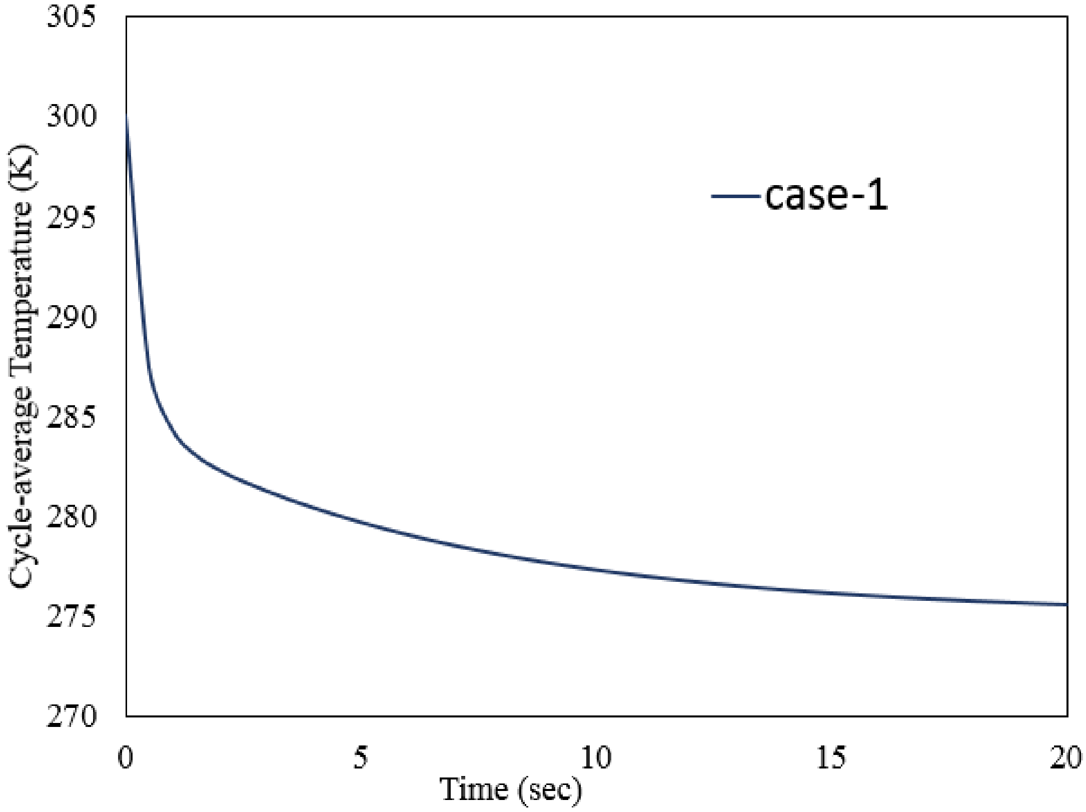


Figure 4.4: The variation of cycle average temperature of CHX surface with respect to time.

In the above Figure 4. 4 CHX temperature initially was at 300 K. The temperature of CHX in 0.5 second to 1 second goes down 284 K. In each cycle the total heat energy absorb from the chamber and rejected to environment and remaining heat supply to pulse tube in ITPTR. The same process are repeated for each cycle. Thus the temperature of CHX surface decreasing rapidly with time. At last it found the temperature of CHX decreases 275 K till 20 sec but after that there is no great variation in temperature of CHX.

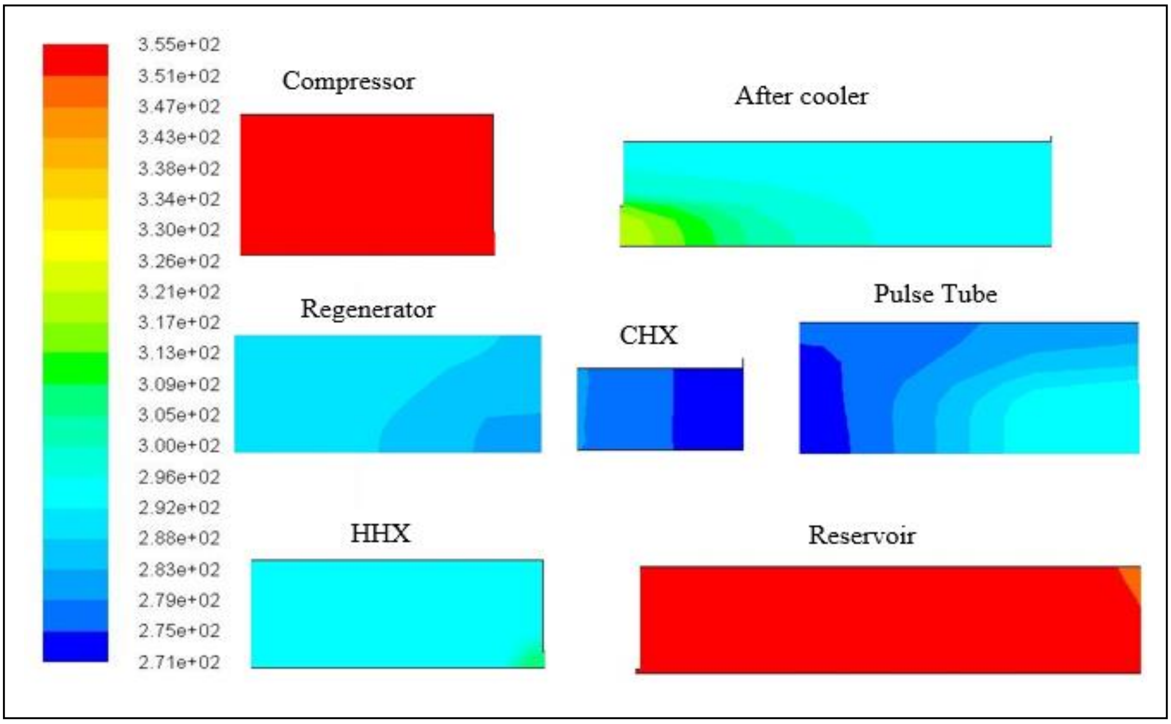


Figure 4.5: Temperature contour using present model with geometry 2 Case-1.

The above Figure 4.5 shows contour of the whole ITPTR and clearly shown lowest temperature at CHX.

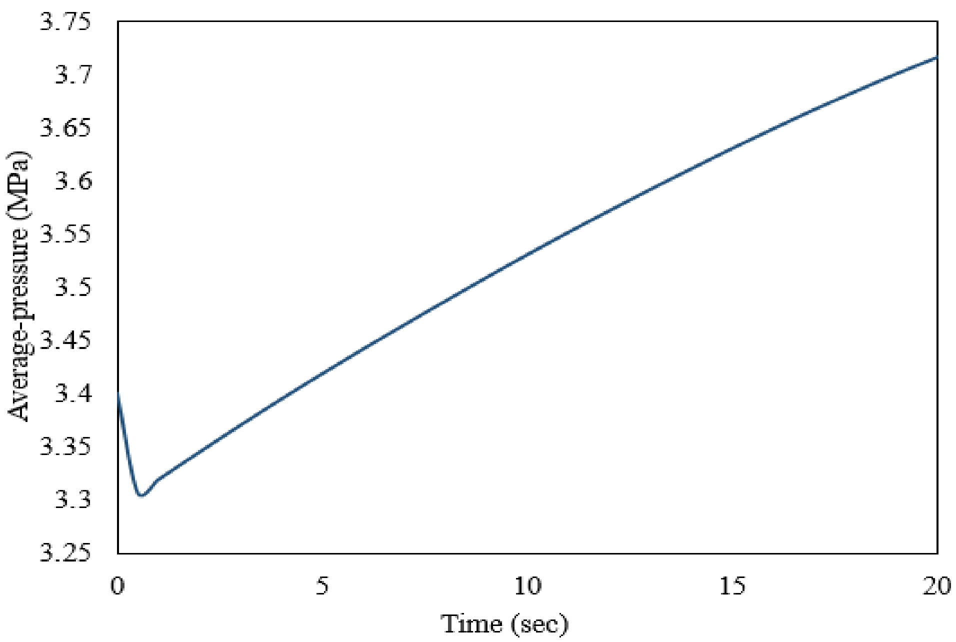


Figure 4.6: The Variation of the average pressure of CHX with respect to time for case 1.

In the above Figure 4.6 shown the variation of average pressure of CHX initially decreases and goes down 3.3 MPa in time of 0.5 second but after 0.5 second the pressure of the CHX increases very rapidly with the respect to the time. The decreasing till 0.5 second and after increasing greatly with time it is due to initially the pressure in compressor was atmospheric pressure but after moving the piston the pressure goes up and thus the at the time 0.5 second the amplitude of mass flow rate in CHX was very large and after 0.5 second the mass flow rate decreasing and pressure increasing. At time 20 second it shows the 3.7 MPa.

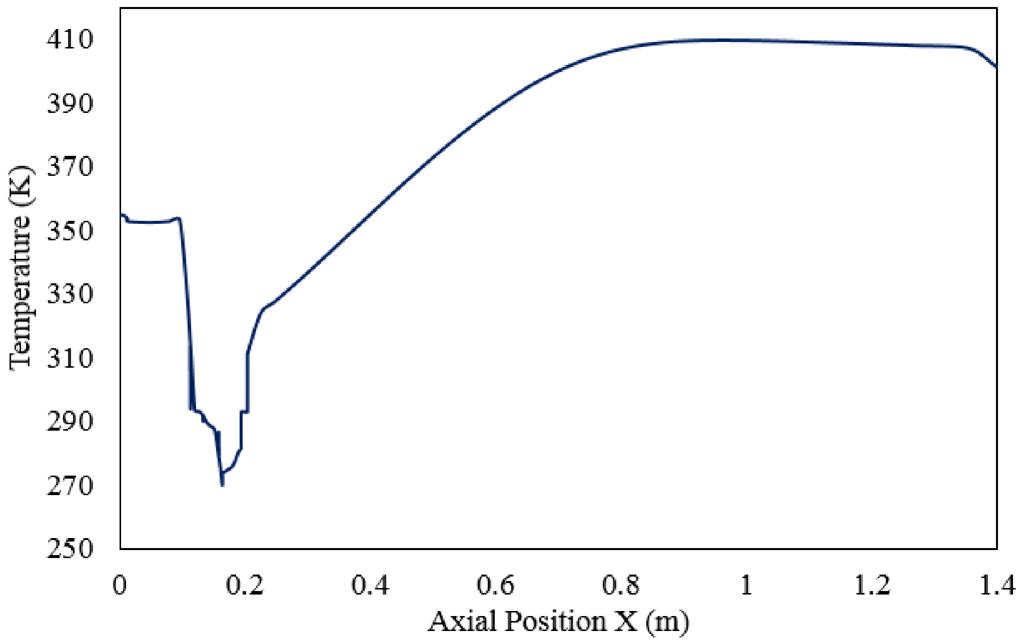


Figure 4.7: Axial temperature variation ITPTR at the 20 second.

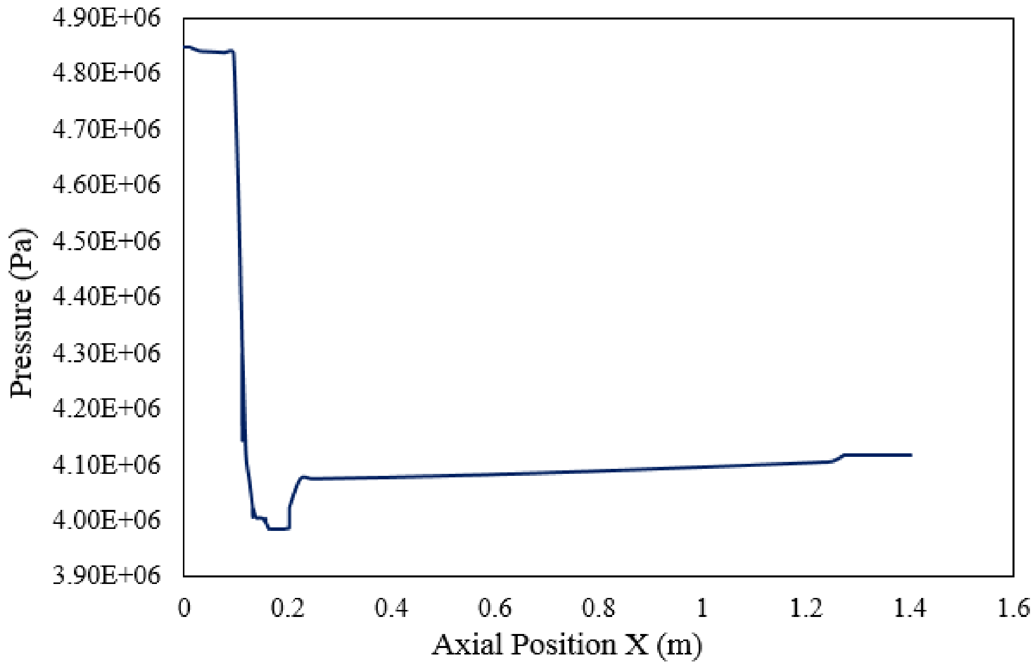


Figure 4.8: Axial Pressure variation of ITPTR at the 20 second.

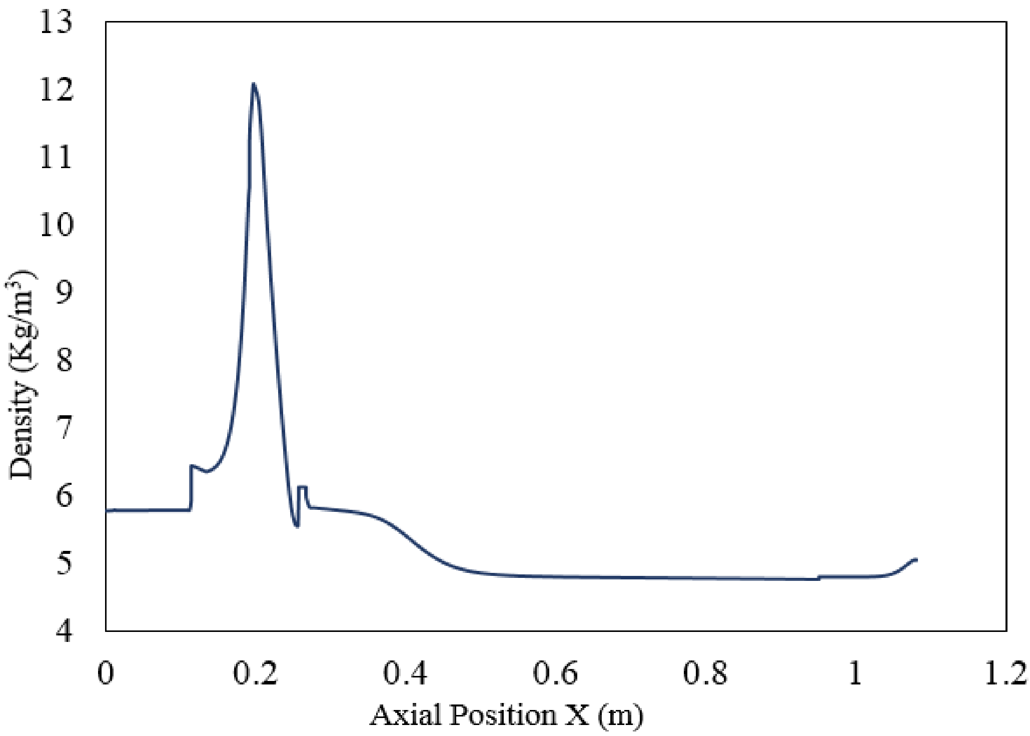


Figure 4.9: Axial Density variation of ITPTR at the 20 second

As shown in Figure 4.7 due to compression and expansion of piston compressor temperature is higher than operating temperature. This extra generated temperature by compressor removed to environment with help of after-cooler. Momentum losses because of the extra source term inside the regenerator. There are two term presence such that Darcy and Forchheimer in which inside porous zone pressure drop is directly depend on velocity. From the above figure 13: absorbed that the minimum temperature at the CHX end is 275 K.

Figure 4.8 shows the variation of pressure inside whole system during the simulation when the piston at IDC inside compressor. Inside the CHX the nature of the Area Weighted Average pressure variation sinusoidal. Compressor has the maximum pressure 4.8 MPa after that the pressure change very rapidly and at CHX shows the minimum pressure 3.97 MPa and it little bit increases with axial position till reservoir pressure 4.1 MPa

Figure 4.9 shows the density variation with axial position, from figure it absorbed that the density of the Helium which use as working fluid of the ITPTR is first increases very high rate and after decreases very rapidly at the CHX. The maximum density found 12 Kg/m^3 at cold heat exchanger. Due to low temperature and pressure at the CHX shows the maximum density.

4.2.1 Case-2: When both the amplitude and volume of the compressor are doubled from the same reference compressor i.e. taken from Cha et al. (2006).

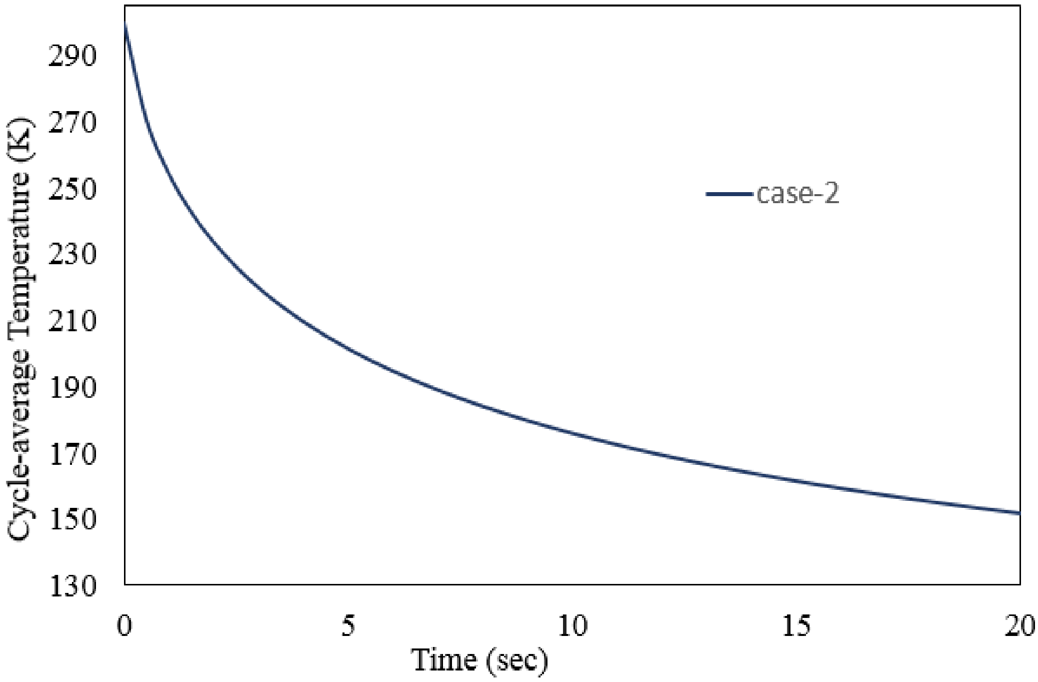


Figure 4.10: The variation of cycle average temperature of CHX surface with respect to time.

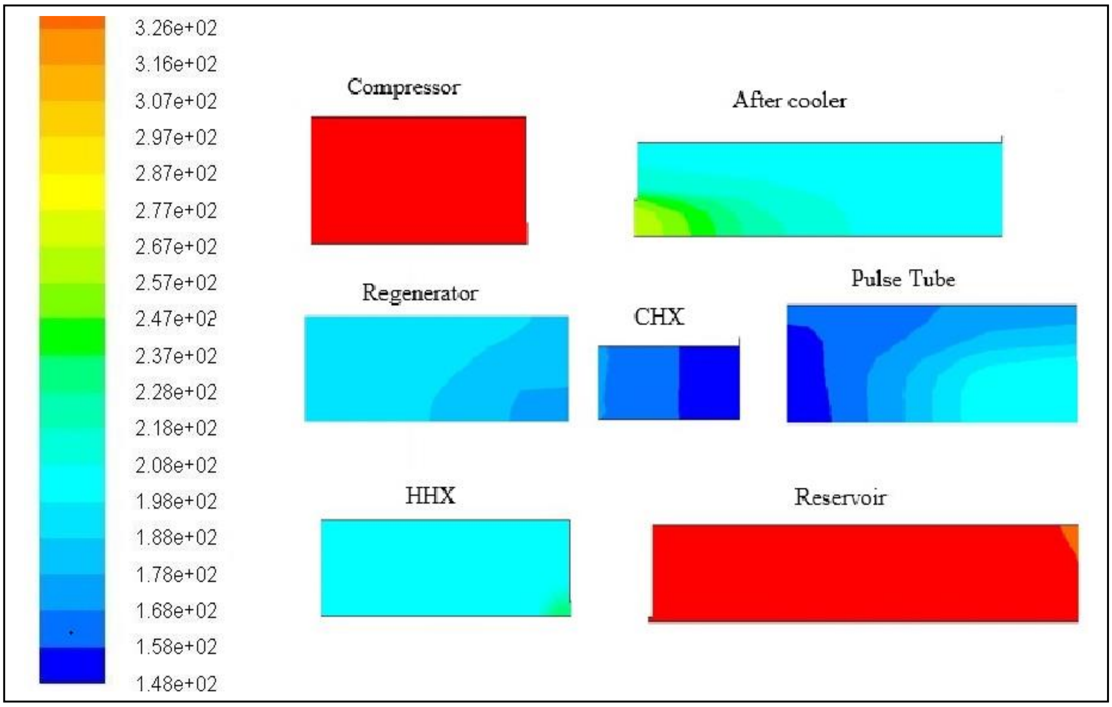


Figure 4.11: Temperature contour using present model with geometry 2 Case-2.

In above Figure 4.10 By increasing volume and amplitude for compressor the cycle average temperature of CHX is rapidly decreasing with the time. The initial temperature of CHX was 300 K but after 20 second it decreases 151 K. The Figure 4.11 shows the temperature contour for different parts of ITPTR with geometry-2 case-2 using present model.

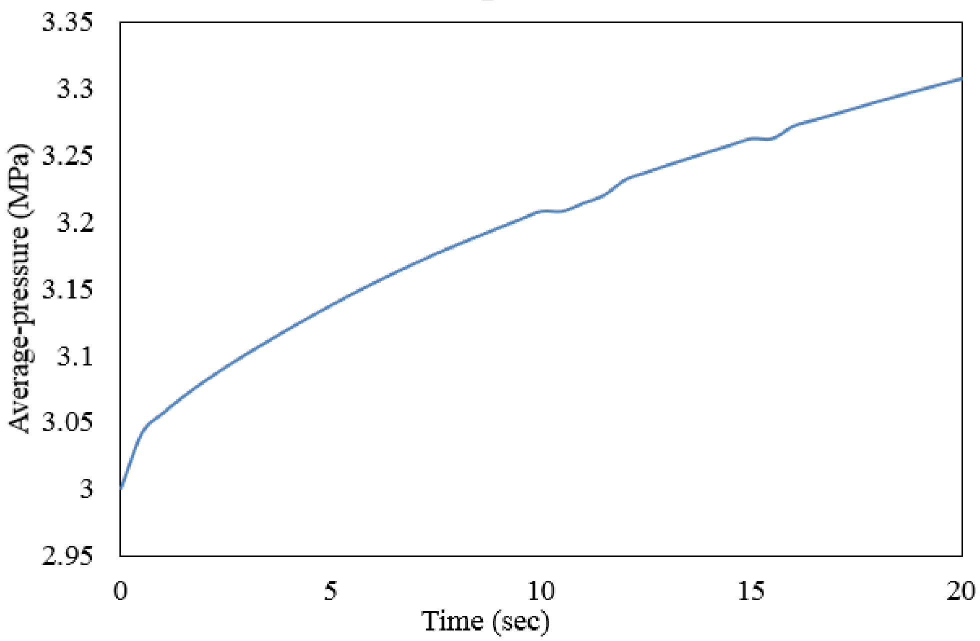


Figure 4.12: The variation of the average pressure of CHX with respect to time.

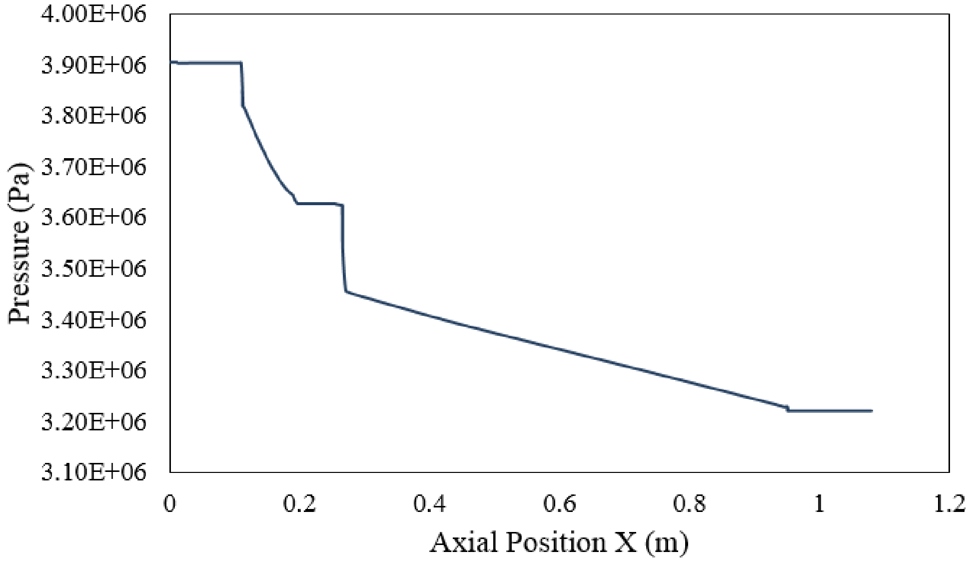


Figure 4.13: Axial Pressure variation from compressor to reservoir at the 20 second.

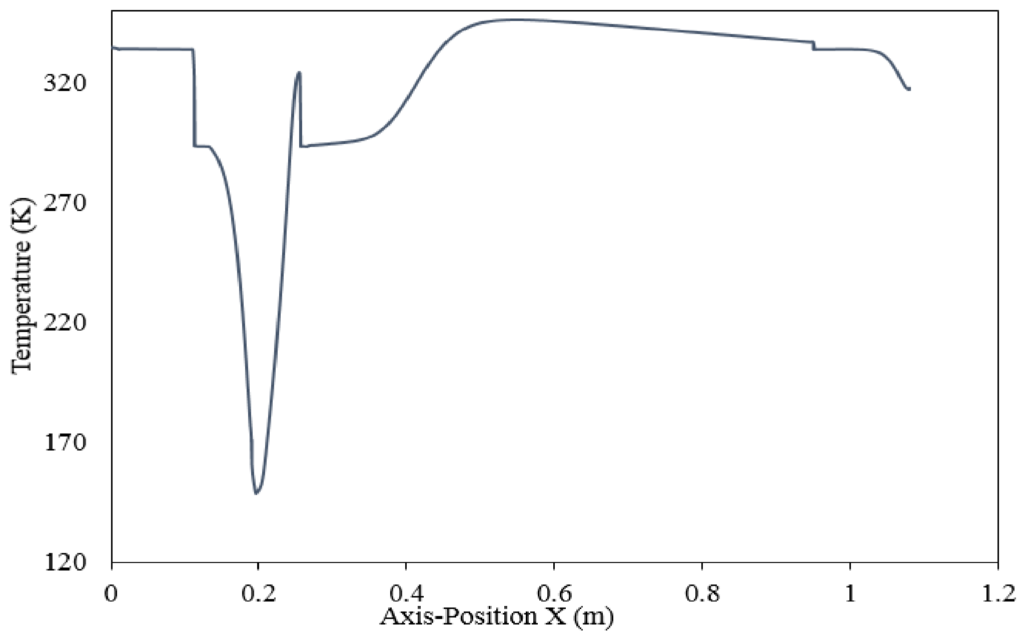


Figure 4.14: Axial temperature variation from compressor to reservoir at the 20 second.

As shown in Figure 4.12 there is no fluctuation of average pressure of CHX at the initial time at 0.5 second as case-1 and it gradually increases with respect to time. This is due to double swept volume by the piston in compressor and also the amplitude of piston movement double. At the time 0 second 3 MPa average pressure and it rapidly increases at 20 second 3.3 MPa.

Figure 4.13 shows the Axial Pressure variation from compressor to reservoir at the 20 second. The highest pressure at the compressor 3.9 MPa and the lowest at inertance tube 3.2 MPa. This curve shows the satisfactory result to obtain the lowest temperature at CHX.

Figure 4.14 shows the lowest temperature at the cold heat exchanger which want to desirable for designing and modeling for ITPTR. The lowest temperature at CHX is 151 K in 20 second. It means the double volume of working fluid and double amplitude of compressor is very useful to achieve early lower temperature of CHX and the density variation with axis position is similar to the case-1.

4.3 Comparison between Cha et al. (2006) model and Case-1

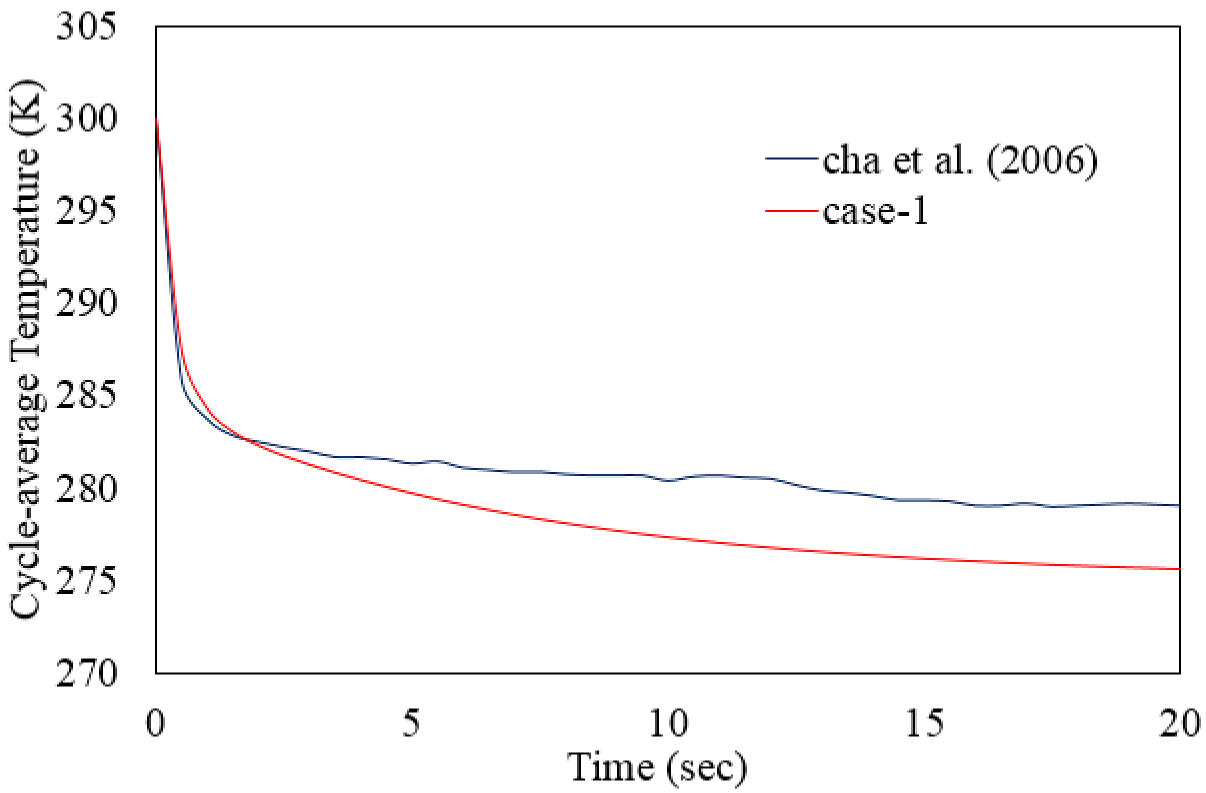


Figure 4.15: Comparison of temperature variation between Cha et al (2006) and case-1

From above Figure 4.15 shows, modeling and computational simulation has been done for case-1 as given dimension and compare it the model of cha et al. (2006) model. At the time 20 sec the cycle-average temperature with respect to time for cha et al (2006) and case-1 are shows 279 K and 275 K respectively. That there is no great variation of the cycle-average temperature with respect to time for case-1 with compare to cha et al (2006) model. Thus the effect of the double amplitude and same volume of case-1 is not give the satisfactory result to design the inertance tube pulse tube refrigerator.

4.4 Comparison between Cha et al. (2006) model and Case-2

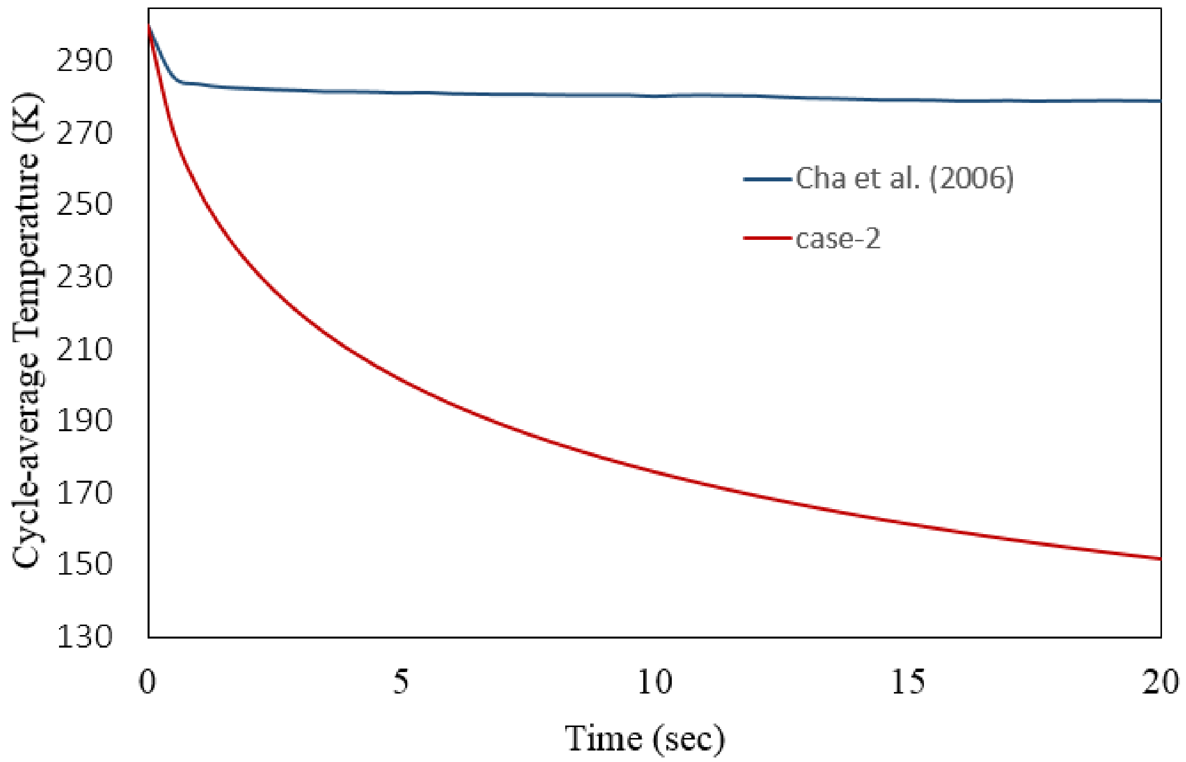


Figure 4.162: Comparison of temperature variation between Cha et al (2006) and case-2.

From above Figure 4.16 shows, modeling and computational simulation has been done for case-2 as given dimension and compare it the model of cha et al. (2006) model. At the time 20 sec the cycle-average temperature with respect to time for cha et al (2006) and case-2 are shows 151 K and 279 K respectively. The cycle-average temperature decreasing with respect to time for case-2 with compare to cha et al (2006) model. Thus the effect of the double amplitude and double volume of case-2 contain the satisfactory result to design the inertance tube pulse tube refrigerator.

4.5 Comparison between Case-1 and Case-2

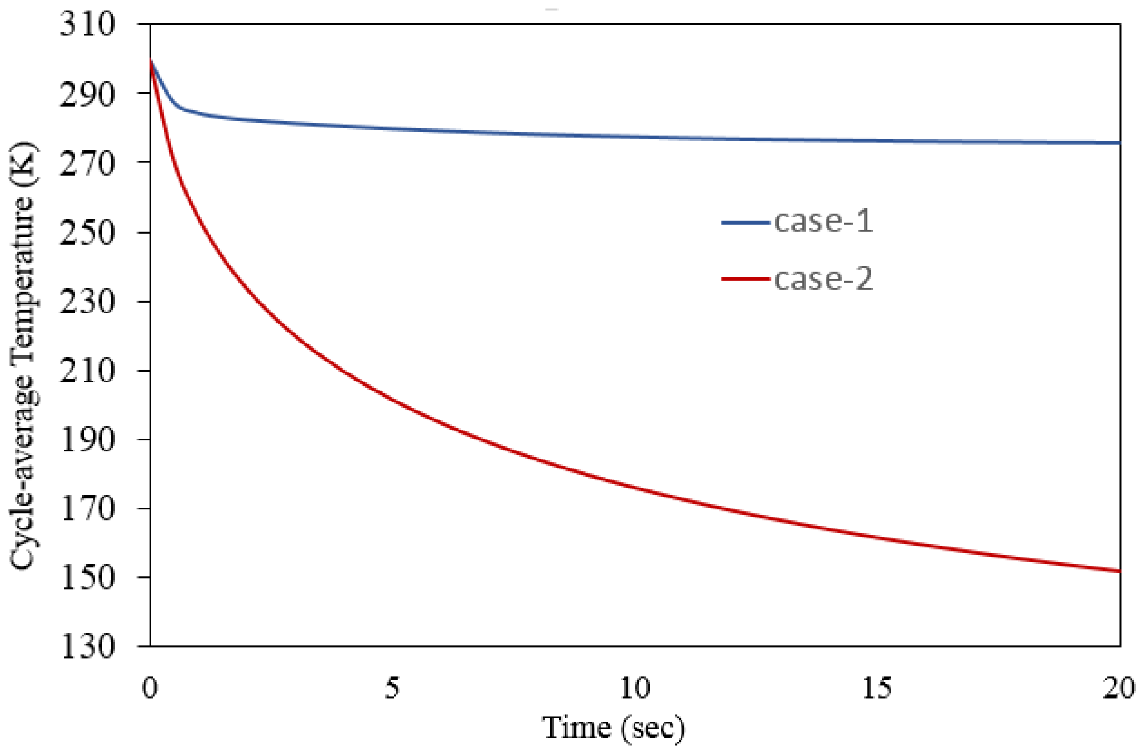


Figure 4.17: Comparison of temperature variation between case-1 and case-2

Modeling and computational simulation has been done for both case-1 and case-2 with given dimension and compare both each other. In the Figure 4.17 at the time 20 sec the cycle-average temperature with respect to time for case-1 and case-2 are shows 275 K and 151 K respectively. The average temperature for case 2 decreases early compare to the case-1. For case-1 and case-2 variation of average pressure with respect to time 3.3 MPa to 3.7 MPa and 3 MPa to 3.3 MPa respectively for 20 second. There is identical variation of density of working fluid Helium in both cases with respect to axis position. Thus all result shows that case-2 (both the amplitude and volume of the compressor are doubled from the same reference compressor i.e. taken from Cha et al. (2006).) has lower fluctuation of average pressure as well as has higher heat transfer capacity to cold end to warm end to achieve lowest temperature at the CHX compare to the case-1.

CHAPTER 5

CONCLUSION AND FUTURE SCOPE

5. CONCLUSION AND FUTURE SCOPE

Here two different inertance tube pulse tube refrigerator (ITPTR) of different dimensions and but same boundary condition were modeled numerically simulated to prediction of performance of ITPTR using the ANSYS. The finite volume method were used for numerical simulations. The first case was the amplitude doubled but the swept volume of compressor remain same as a reference compressor i.e. taken from Cha et al. (2006) and the second case was both the amplitude and volume of the compressor are doubled from the same reference compressor. From analysis of the result it has been concluded that.

- In order to obtain the accurate model and solution method validation test was demonstrated.
- The two different ITPTR were compared with the Cha et al model.
- The temperature of CHX for case-2 was 151 K whereas Cha et al (2006) model and for case-1 model reported 279 K and 275 K temperature respectively at 20 second.
- Thus the case-2 demonstrate the temperature of cold heat exchanger (CHX) was early cool down with respect to time as compare to other two model Cha et al model and case-1 due to increasing the amplitude as well as volume.
- The variation of the average pressure with respect to time for case-2 is lower than the case-1. Thus the vibration inside the case-2 system also lower than the case-1 model which is desirable to designing for ITPTR.
- By the optimum combination of amplitude and volume of compressor in ITPTR the minimum temperature at CHX as well the good temperature decreasing rate can be achieved.

Future scope:

1. The all cases can also done by experimentally and investigate both simulation and experiment result.

2. Methodology can be developed to find out the optimum combination of amplitude and swept-volume in the compressor of ITPTR to get the minimum temperature at CHX as well good temperature decreasing rate.

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